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Assessment of HC-290 Charge Reduction in Split Type Air Conditioner

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Abstract

Global warming is at the forefront to drive and develop method or equipment to avoid climate change. Kyoto protocol guides air conditioning sector to avoid use of HFCs by adopting an alternative refrigerant. HC-290, a potential alternative refrigerant, possesses negligible Ozone Depletion Potential (ODP) and a minimal Global Warming Potential (GWP). However, it is highly flammable refrigerant imposing limitation on the charge quantity that can be used in the Split Air Conditioner (SAC). Present study describes Vapor Compression Cycle (VCC) analysis for the refrigerant charge reduction by variation of some of the geometric parameters for the condenser and it also describes possible ways to reduce charge by variation to condenser without change in system process parameters of SAC. By optimizing condenser design through parametric analysis and simulation, this work explores refrigerant charge reduction in an HC-290 split air conditioner. A mathematically designed vertical tube condenser was used, and the system maintained a cooling capacity of 5100 kW and an EER of 3.85% while achieving a 24% reduction from the baseline 330g charge. The main cause of charge decrease, according to the findings, is condenser change, while expansion device sizing also affects the overall charge mass.

Major Findings: *By vertical tube condenser it is possible to save refrigerant charge in SAC. There is no impact on the cooling capacity and EER for the similar capacity of SAC.*

Keywords: *Air Conditioner, Charge Reduction, Falling Film Condenser, HC 290, Natural Refrigerant*

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1.0 Introduction

Global warming has directed the world to discuss and sign Kyoto protocol. The resolution restricts the use of HFCs as the refrigerant in the refrigeration and air conditioning sector. The agreement to the Kyoto Protocol has avoided more than 135 billion tonnes CO_2 emissions through phasing out their production and consumption. As per the Kigali Amendment to the Kyoto Protocol, the phase out started in 2019 for developed countries. Second stage is in 2024 for group 1 of non-articles 5 countries and 2029 is for the second group of non-articles 5 countries. These years will freeze the baseline for group countries. These restrictions are the challenge and an opportunity for researchers and industry to develop alternatives for conventional refrigerants. Therefore, industries worldwide are investigating alternatives that possess zero Ozone Depletion Potential, low Global Warming Potential, low toxicity, and high chemical stability as refrigerants. Additional characteristics such as appropriate thermodynamic, physical, and chemical characteristics, substance suitability, and cost-effectiveness are also significant.

Strict restrictions on the manufacturing and use of High Global Warming Potential (HGWP) refrigerants including R22, R134a, R410A, and R32 have been

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enforced in response by international accords, most notably the Kigali Amendment. In addition to lowering the negative effects of refrigerants on the environment, these regulations offer a chance to include energy-efficient STAC technology[1]. Reduced CO_2 emissions, enhanced techno-economic synergy, and less production of hazardous synthetic byproducts could be the outcomes of the combined effect. Because of better thermophysical qualities, R1270, R600a, and R290 hydrocarbon refrigerants are now considered viable replacements to HGWP refrigerants[2]. Despite being categorized as A3 (very flammable) by ASHRAE 34, these natural refrigerants have significant environmental benefits, such as nontoxicity, 0% ozone depletion potential, and ultralow global warming potential. This based on the literature[3-4] HC-290 is a feasible alternative. HC-290 exhibits a zero-ozone depletion potential and possesses a remarkably low global warming potential, characterizing it as a naturally occurring fluid. It does not influence climate in any other way, though it is classified as an organic volatile compound. According to ASHRAE 34-2016, HC-290 classified as A3. It has low toxicity but is highly flammable. The HC-290 charge quantity must be decreased in the system to remain below the lesser volatile limit as outlined in the European Standard 378. The objective of this paper is to study possible alternatives to reduce refrigerant charge quantity below the lower flammable limit. Refrigerant charge reduction is achieved without affecting the performance of the air conditioner, i.e. the cooling capacity, COP are maintained similar as the base case for the same operating conditions.

2.0 Studies in Charge Reduction

Poggi *et al.,*[5] examined several strategies to diminish charge in refrigeration systems, including the selection of alternative refrigerants, pipe sizing, sizing and loading of liquid receivers, implementation of secondary refrigeration, appropriate heat exchanger selection, and modifications to compressors and expansion valves. For direct expansion evaporators and condensers, the most straightforward method to decrease the charge is to flatten the tubes. Flattened tubes possess a superior interior surface-to-cross-sectional ratio in comparison to the original spherical tubes. Flattening tubes concurrently enhances the heat transfer coefficient and the pressure drop for a specified refrigerant mass flow rate during both boiling[6] or condensing[7]. The mass flux will increase, resulting in a transition from stratified flow to annular flow at lower mass fluxes in the evaporator. The circular flow has a superior heat transfer coefficient compared to stratified flow, hence enhancing heat transfer efficiency[6]. In tiny heat exchangers, such as microchannel heat exchangers, the refrigerant quantity is diminished to approximately 10 grams per kilowatt when headers are properly engineered[8]. The additional advantages of employing microchannel heat exchangers encompass reduced dimensions and less weight.

Using a microchannel condenser[9] for a 3.2 kW Split Air Conditioner (SAC) charge reduction to 190 g for HC-290 refrigerant was achieved. The arrangement of header and refrigerant tubes that they made has a header at one of the tubes and it is eliminated at the other end. Reduced condenser tube diameter and increased capillary tube length study[10] resulted in 320 g of HC-290 refrigerant for 5.2 kW SAC can show similar performance as the base line or even slightly better. Falling film condenser with higher capacity compressor[11] for HC-290, performance of the 2.6 kW air conditioner was improved by having slightly improved cooling capacity. Three different expansion devices used as the throttling device in the vapor compression cycle[12]. In conclusion, it is optimum to have a thermal expansion device in cycle. As sustainability and energy conservation gain greater attention globally, there is a growing need for air conditioning systems to be more ecologically friendly and efficient[13]. Traditionally, HVAC performance has been predicted using analytical solutions or empirical correlations, which may not be adequate to describe the intricate relationships between the many different elements that affect system performance[14].

In current study design modifications to the existing 5100 W capacity SAC are studied. Considering techniques presented by researchers few alternatives are studied from the point of view of refrigerant charge reduction. The conventional SAC has a condenser which is a horizontal tube condenser. In place of this condenser, a vertical tube condenser is considered. The basis and the modifications in the design are described in the following section.

3.0 Refrigerant Charge Reduction Study

Typical AC with nominal capacity of 5100 W was selected for the study. Table 1 compiles parameters for Compressor, Condenser, Evaporator, Expansion device used in the split air conditioner. The specifications of condenser, evaporator, expansion device and process parameters are tabulated here under.

Table 1. Compressor parameters

VCC component	Description (1.5 T or 5100 W) Split AC unit
Compressor	21 $cm^{3*}rev^{-1}$, 925 W, 480 – 7200 RPM (8 – 120 Hz)
Condenser	Copper Tube 5 mm OD, 4.5 mm ID, L 756 mm, Fin spacing FPI 21
Evaporator	Copper Tube 7 mm OD, 6.5 mm ID, L 715 mm, Fin spacing 1.4 mm
Capillary Tube	1 Tube, 1.91 mm Dia, 500 mm Length

Table 2. Condenser parameters

Sr. No.	Parameter	Details
1	Tube material	Copper
2	Tube type	Inner-Grooved tube
3	Tube OD (mm)	5
4	Tube ID (mm)	4.5
5	Tube spacing X_t (mm)	19.05
6	Tube spacing X_l (mm)	11.2
7	Number of rows	2
8	Number of tubes per Row	26
9	Single tube length (mm) along ODU length	1st Row-L 756, 2nd Row-L756
10	Fin type	Slit
11	Fin material	Aluminum
12	Fin per inch (FPI)	FPI 21
13	Fin thickness (mm)	0.105

Table 3. Evaporator parameters

Sr. No.	Parameter	5 kW Value
1	Tube material	Copper
2	Tube type	Inner-Grooved
3	Tube OD (mm)	7
4	Tube ID (mm)	6.5
5	Tube spacing X_t (mm)	21
6	Tube spacing X_l (mm)	12.5
7	No of rows	2
8	No of tubes per row	16
9	Single tube length (mm) along IDU length	715
10	Fin type	Slit
11	Fin material	Aluminum
12	Fin spacing (mm)	1.4 mm
13	Fin thickness (mm)	0.1

These components specifications from Table 1 to Table 4 when used in a simulation software IMST ART[15] the cycle parameters are derived. These cycle parameters

Table 4. Capillary tube and tubing data

Sr. No.	Parameter	5.2 kW Value
1	No of capillary tubes	1
2	Inner diameter	1.91
3	Length (mm)	500
4	Condensing temperature (^{0}C)	51.5
5	Condenser exit subcooling (^{0}C)	42.3
6	Evaporator temperature(^{0}C)	9.1
7	Evaporator exit superheat (^{0}C)	12.7
8	Refrigerant charge used in the system (g)	330
9	Refrigerant flow rate (kg/hr)	11.23

System line dimensions (mm OD and mm Length)		
10	Suction port to suction valve	OD 12.7 / L865
11	Discharge port to discharge inlet header	OD 9.52 / L648
12	Indoor coil to compressor inlet	OD 12.7 / L4365
13	Compressor outlet to outdoor coil	OD 9.52 / L648

Table 5. Cycle parameters by ART simulation

Parameters	100% capacity	50% capacity
EER (W/W)	3.16	4.22
Cooling capacity (kW)	5.001	2.503
Total Power input (kW)	1.58	0.594
Discharge Temp. (^{0}C)	63.12	49.54
Refrigerant flow rate (kg/h)	63.72	39.6
Pressure ratio	2.92	--
Evaporating temp (^{0}C)	9.46	--
Condensing temp (^{0}C)	51.5	--
Subcooling (K)	8.3	--
Superheat (K)	2.39	--
Total charge (g)	230.9	230.9
Charge in condenser(g)	120.7	62.32

are listed in Table 5. The model uses REFPROP v7 for thermodynamic and transport property parameters. Using this simulation software, it is possible to build a horizontal tube condenser. To do the parametric study of condenser and capillary tube the cycle parameters listed in Table 5 are considered as base. Comparison is done for every parameter for the refrigerant charge, cooling capacity and EER with the base case.

3.1 Design of Experiments (DOE)

Based on the literature survey it is inferred that design of condenser and capillary tube parameters have the maximum impact on the EER and refrigerant charge for same cooling capacity. This is because these parameters can control the evaporator pressure and temperature. As the evaporator pressure is slightly reduced, it can improve the cooling capacity or in other way can reduce the charge for the same cooling requirement. By performing the DOE, the refrigerant charge is observed verifying the parameters such as EER, Cooling capacity for rated and part load output. In each DOE one parameter for condenser or capillary tube is varied and results are noted. Various DOEs are as follows.

For condenser parameters varied are tube diameter, tube thickness, tube fin pitch, tube transverse pitch, air velocity. For the capillary tube parameters varied are capillary tube diameter, capillary length, number of capillaries with reduced diameter. Results of all these DOEs are listed and compared.

3.1.1 Condenser Parameters

Figure 1 shows the EER, Figure 2 shows cooling capacity and Figure 3 shows refrigerant charge comparison for various condenser parameter DOEs.

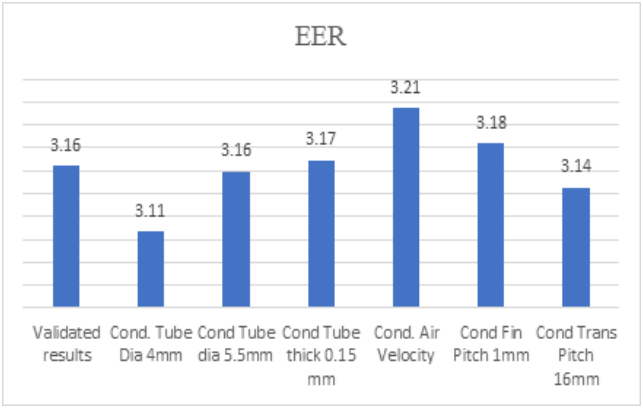


Figure 1. Simulation comparison of condenser parameters for EER.

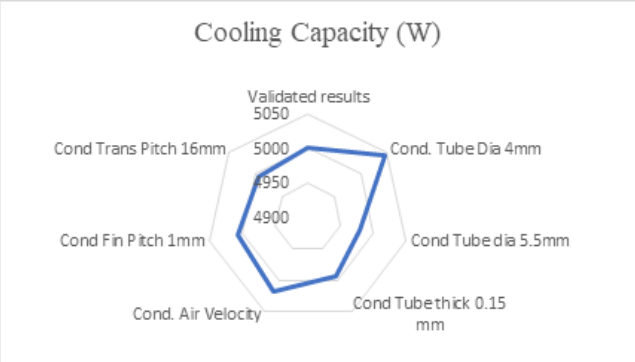


Figure 2. Simulation comparison of condenser parameters for cooling capacity.

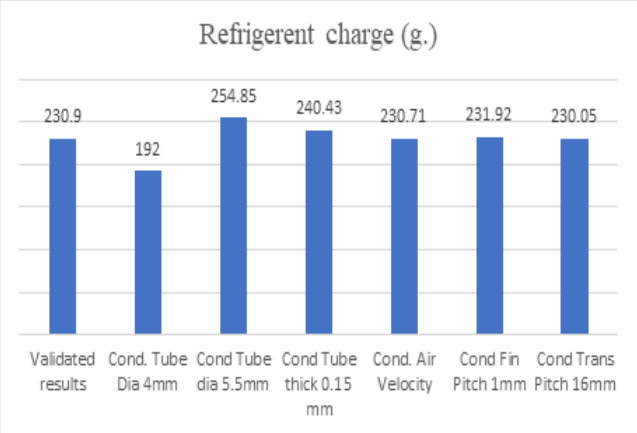


Figure 3. Simulation comparison of condenser parameters for refrigerant charge.

Few Observations

- For condenser tube diameter of 4 mm the charge is reduced (around 40 g) in comparison with base case, but the EER has dropped by (0.05). Cooling capacity has increased by 45 W.
- Charge quantity is least for 4 mm condenser tube case. For all other DOEs for condenser parameters charge quantity has increased or is nearly same as base case. However, the EER is lowest compared to other DOEs. Pressure drop is also increased substantially, due to which EER is poor for 4 mm condenser tube case.
- Increasing the condenser tube diameter as expected refrigerant charge is increased but the EER is same as the base case, and cooling capacity is reduced.

3.12 Capillary Parameters

Comparison for capillary tube parameters DOE cases are presented in Figures 4 to Figures 6.

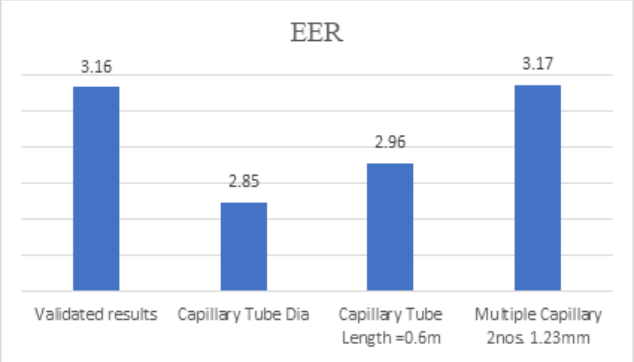


Figure 4. Simulation comparison of capillary tube parameters for EER.

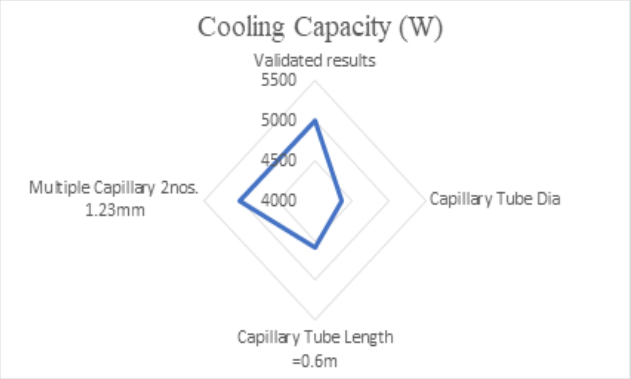


Figure 5. Simulation comparison of capillary tube parameters for cooling capacity.

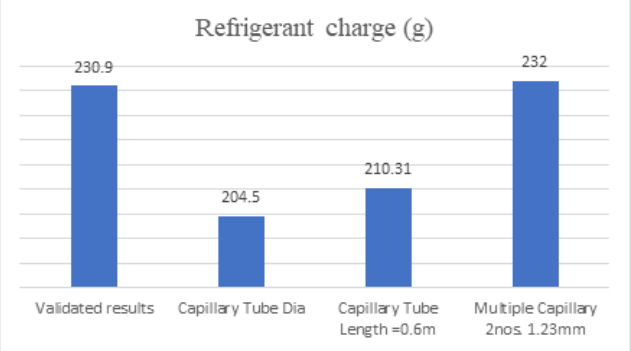


Figure 6. Simulation comparison of condenser parameters for refrigerant charge.

Few Observations

- When the Capillary tube parameters are varied the charge is reduced by around 20 g, but there was reduction in cooling capacity by around 50W.
- For multiple capillary tubes EER has increased but the charge is also increased.

4.0 Simulation of Vertical Tube Condenser

The simulation tool used for the cycle simulation is able of designing conventional condenser. To simulate vertical tube condenser with same boundary condition i.e., inlet and outlet condition, MATLAB model is made. The developed mathematical model is used to solve a thermal sizing problem. Geometry of the condenser is considered as input for the model and cooling capacity of condenser is calculated. For this following methodology is used.

4.1 Basis for Condenser Modelling

- Condensation divided in three sections – De superheating, two phase flow and subcooling
- Boundary conditions as per model
- Approach – Thermal capacity sizing Flow chart for the mathematical model is shown in Figure 7.

Following correlations are used for the mathematical model of vertical tube condenser.

- De-superheating section

To calculate heat transfer coefficient, it is required to calculate Reynolds number, Prandtl number, Nusselt number for the fluid used for the heat transfer. After that using the standard co-relation, the heat transfer co-efficient is calculated. For calculating heat transfer coefficient on air side, the HTRI recommended correlation is used to calculate the air side heat transfer coefficient. Then the overall heat transfer coefficient for the de-superheating section is calculated.

HTRI (16) recommended correlation for the heat transfer is as follows,

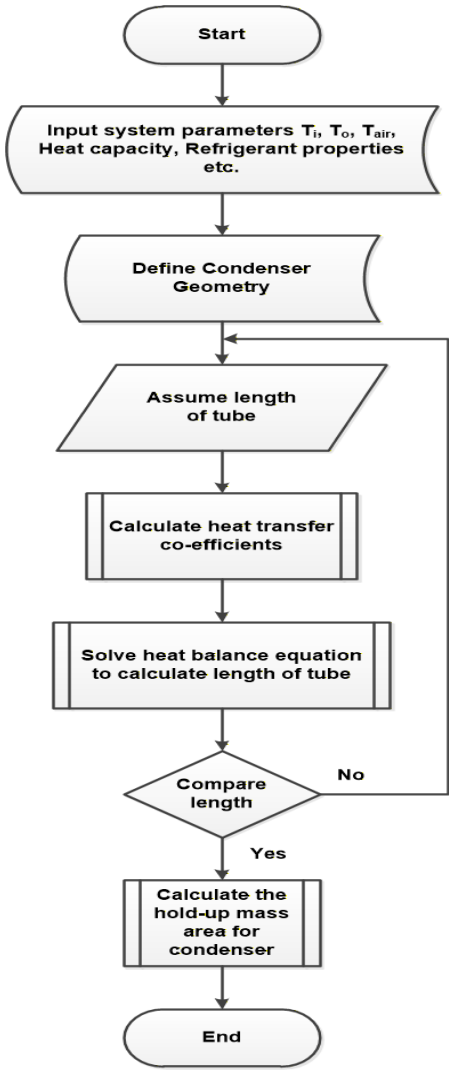


Figure 7. Flow chart for MATLAB condenser calculation.

$$Nu = 0.38 \times Re^{0.6} \times Pr^{1/3} \times \left(A_{Tot}/A_o\right)^{-0.15} \quad (1)$$

where,

A_{Tot}= Total External surface area of finned tube

$A_o = \pi D_r L$=Total external surface area of root tube

where

$$Nu = h_o \times D_r/k \quad (2)$$

Based on this then the amount of heat transferred in the de-superheating section is calculated. Here the conduction resistance of the condenser tube is neglected.

$$\frac{1}{U * A_o} = \frac{1}{h_r A_i} + \frac{1}{h_a A_o} \quad (3)$$

$$Q_1 = U_1 * Ao * \Delta T \quad (4)$$

- Two phase flow section

This is the process in the condensation where maximum heat transfer takes place. Due to the falling film phenomena the heat transfer from this section is observed to be highest. In the horizontal tube condenser, the liquid retention progressively escalates along the tube. The majority of the refrigerant retention occurs in the two-phase and subcooled regions. In a vertical tube, the refrigerant flows downhill, aligned with the direction of gravity. So, as the refrigerant gets cooled and phase change occurs. The liquid film is formed along the tube wall, due to gravitational force the film thickness is restricted, and heat transfer rate is always enhanced.

The condensed liquid film disseminates along the tube wall and descends, while its thickness diminishes due to gravity and the shear stress between the gas and liquid. Consequently, the liquid film does not exceed a certain threshold, facilitating heat transfer. This approach can evidently diminish the buildup of refrigerant in the condenser and the overall refrigerant charge of the system.

Correlation used for the heat transfer coefficient on refrigerant side recommended by (11)R290 has flammability. There is necessary to improve the STAC design to reduce the refrigerant charge for the safety use. This work proposes a novel falling film condensing technology to address this problem. The novel condenser was setup on a STAC and tested in the standard psychrometric chamber. Experimental results show that using falling film condenser significantly reduces the refrigerant charge. The best performance is obtained with 290 g of R290 charge, for which the cooling capacity is 2690 W and the coefficient of performance (COP.

$$h_2 = \left[\frac{g \times r \times \rho_l^2 \times \lambda_l^3}{4 \times \mu_l \times (T_s - T_w) \times Z}\right]^{0.25} \quad (5)$$

r=Latent heat (J/kg K)
λ=Thermal conductivity (W/m K)
T_s=Saturation temperature (K)
Z=Characteristic dimension (m)

Using the same equation of overall heat transfer the overall heat transfer coefficient is calculated and the cooling capacity on the two-phase section is calculated

$$Q_2 = U_2 * Ao * \Delta T \quad (6)$$

Subcooling section

This section is a part of condenser where the heat transfer takes place between liquid refrigerant and the surrounding air. Due to this the refrigerant leaving the condenser becomes sub-cooled. This section is again a single-phase section and using all liquid refrigerant properties and same correlations as used in the de-superheating section the heat transfer is calculated for this section.

- Overall cooling capacity of the condenser

By adding all the heat transfer in three sections together total capacity of the condenser is calculated. In the current simulation the characteristic length was initially assumed as 1 fin distance and then for every iteration it was increased by a fin pitch. The length of each cooling segment of the condenser is computed and subsequently aggregated to determine the total length of the condenser tube. As mentioned in the flow chart the iterative calculation concluded the results of the mathematical model. The geometry of the condenser was defined based on the results of the mathematical model. Based on this geometry, total charge in the condenser is calculated. Adding the charge to the rest of the cycle component total charge for VCC operation is derived. Following table shows the comparative of the vertical tube condenser with the simulated horizontal tube condenser.

5.0 Experimentation

Using the simulation tool, it is ensured that the operating boundary conditions are maintained. By that the overall heat transfer capacity, refrigerant mass flow rate, operating pressure etc. parameters of the VCC cycle are maintained.



Figure 8. Image of falling film condenser.

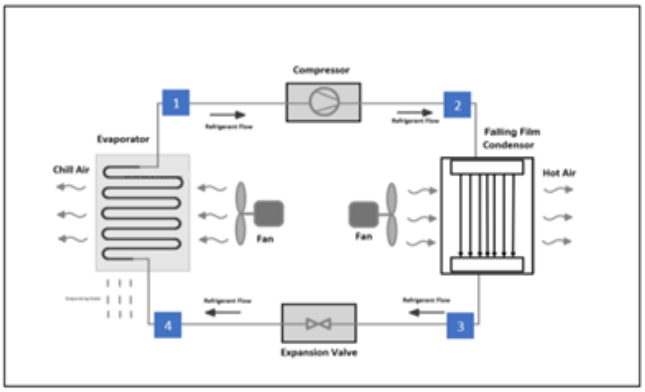


Figure 9. Experimental set up of falling film condenser.

So, the condenser tube with the least charge will be the best suited design. For the validation of the falling film design a test set up with 7 mm tube OD condenser is fabricated. Replacing the existing horizontal tube condenser with the falling film condenser, split air conditioner is operated to test. Schematic of the test set up and image of falling film condenser developed is shown in Figure 8.

The SAC selected is a 5100 W cooling capacity air conditioner. The conventional condenser is replaced with the falling film condenser. During the experimentation pressure, temperature is noted for four stages as indicated in Figure 9 These parameters are used to plot the vapor

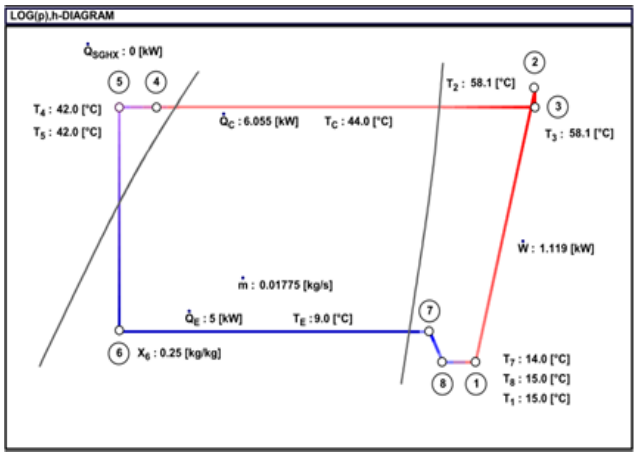


Figure 10. Experimental falling film condenser VCC p-H plot.

compression cycle on the p-H chart. This p-H chart is presented in Figure 10.

Operating parameters and performance parameters of the experimental set up with falling film condenser having 7 mm tube OD, are comparable to the base case and simulated case. As can be seen from Figure 10 the cooling capacity, condenser cooling capacity, refrigerant mass flow rate, evaporator temperature etc. these parameters match within 5 %. During this experiment the refrigerant was charged up to 6.89 bar when the system was in static condition.

5.1 Experimental Uncertainty

By analyzing the precision of the calibrated transducers used in the tests, the experimental results for air velocity, cooling capacity, and EER were evaluated for unpredictability. The air and refrigerant properties are expected to be within ±1% of accuracy. Air flow's relative standard uncertainty is estimated to be within 1.0%. For cooling capacity, the relative standard uncertainty falls between 3.76% and 4.47%. The EER ranges from 4.47% to 5.17%.

The setup was operated and the pressure and temperature were noted for all the trails. The main motto of operating the test set up was to verify the repeated cooling capacity being achieved and the corresponding changes to the other parameters. The set up was run for mainly full capacity of the unit. It was observed that the

Table 6. Instrument accuracy

Sensor	Test Device	Accuracy	Full scale
Temperature	PT 100	± 0.2 ^{0}C	-50 -150 ^{0}C
Pressure Transducer	Omega Pressure gauge	± 0.25 % of full scale	0 – 1 MPa / 0-2.5 MPa
Energy meter	WT230	± 0.2 % of full scale	7500 W
Anemometer		± 0.002 m/s	0-10 m/s
Digital psychrometer	METRAVI (ET-99TH)	± 0.2 % of full scale	Actual value

change in the operating parameter was very minimal in all the runs. The cooling capacity was almost the same throughout the testing. Parameters noted during the experiment are listed in the result.

6.0 Results and Discussion

6.1 ART Model Simulation Results

The performance of a standard Split Air Conditioner (SAC) using HC-290 refrigerant was established as a baseline, yielding a cooling capacity of 5.1 kW and an Energy Efficiency Ratio (EER) of 3.85%. The initial refrigerant charge for this baseline configuration was recorded at 330 grams.

A mathematically designed vertical tube condenser was integrated into the system through a simulation model developed to maintain these core performance metrics. The results indicate that the vertical tube condenser

Table 7. Condenser simulated cases parameters comparison

Parameter	Base case Experimental	Base case Simulated
Condenser Capacity (W)	6138	6078
Condenser Inlet Temp (^{0}C)	74	70.4
Condenser Outlet Temp (^{0}C)	38.7	39.2
Condenser pressure (kPa)	1771	1652
Condenser Tube length (mm)	756	500
No. of Tubes	52	48
Cooling Capacity (W)	5100	5001
Refrigerant Charge (g)	330	250
Total Heat transfer Area (m^2)	13.16	14.46

Table 8. Condenser geometric parameters

Heat exchanger configuration	Base case	Vertical tube simulation
Tube material	Copper	Copper
Tube type	Inner-grooved tube	Plain
Tube OD (mm)	5	5
Tube ID (mm)	4.5	4.5
Tube spacing X_t (mm)	19.05	19
Tube spacing X_l (mm)	11.2	12
No. of rows	2	2
No. of tubes per rows	26	24
Single tube length (mm)	1st Row-L 756, 2nd Row-L 756	1st Row-L 500, 2nd Row-L 500
Fin type	Slit	Slit
Fin material	Aluminium	Aluminium
Fins per inch (FPI)	FPI 21	FPI 20
Fin thickness (mm)	0.105	0.1

successfully meets the requirements of the overall cycle. Comparative data for the simulation parameters of the base case versus the vertical tube design are presented in Table 7, while the specific geometric parameters of the condensers are compared in Table 8

Geometric optimization played a crucial role in maintaining performance with a reduced charge. As detailed in Table 8, the vertical tube condenser achieved a surface area comparable to the horizontal tube design by reducing the tube length while maintaining a constant tube diameter. Further modifications included the implementation of hexagonal tube fins and an increased fin pitch.

Because the condenser holds the maximum charge in the system, these modifications served as the primary driver for charge reduction. Simulation results confirmed that the refrigerant charge decreased from 330 grams to 250 grams, representing a 24% total reduction. While condenser design was the most impactful factor, observations indicate that expansion device sizing also influences the final refrigerant charge mass quantity. This significant reduction is vital for the safe implementation of HC-290, which is highly flammable despite its environmentally friendly profile of negligible ODP and minimal GWP.

6.2 MATLAB Simulation Results

A parametric study was conducted using a MATLAB falling film simulation to identify variables that impact refrigerant charge without compromising system performance. The simulation utilized an iterative process to compare various condenser tube sizes and ensure the best comparable results for the falling film model. The mathematical model for the proposed vertical tube condenser was designed to match the overall cycle requirements of the baseline system. Table 9 summaries the parameters of condenser geometry for different tube sizes.

Based on the geometry and the internal volume of the condenser, total charge in the condenser is calculated. Adding the refrigerant charge in the rest of the system components total charge in the air conditioner is calculated. The change to the SAC is only the condenser, so the charge difference in the condenser is same as the

Table 9. Comparison of condenser geometry based on tube size

Heat exchanger configuration	**Base case**	**Falling film simulation**	
Tube material	Copper	Copper	Copper
Tube type	Inner-grooved tube	Plain	Plain
Tube OD (mm)	5	5	7
Tube ID (mm)	4.5	4.5	6.5
Tube spacing X_t (mm)	19	19	19
Tube spacing X_l (mm)	12	12	12
Tube length (mm)	500	410	350
No. of rows	2	2	2
No. of tubes per rows	24	24	24
Single tube surface area (m^2)	0.3864	0.3864	0.3288
Charge (g)	250	186	329
Total Heat Capacity (W)	6078	6085.7	6085.7

Table 10. Comparison of condenser geometry and charge prediction

Heat exchanger configuration	Actual SAC data	Base case	Falling film simulation
Tube OD (mm)	5	5	5
Tube ID (mm)	4.5	4.5	4.5
Tube length (mm)	756	500	410
No. of tubes	56	48	48
Charge (g)	234.3	157.4	129

charge difference for the SAC. Results of this analysis are presented in Table 10.

The simulation model developed for the falling film condenser is meeting the overall cycle performance parameters as compared to base case condenser. Therefore, replacing the conventional condenser with falling film condenser, it is possible to reduce the refrigerant charge in the condenser.

Falling film condenser design matches closely with the inlet and outlet boundary condition for the air conditioner system. In addition, it has comparable cooling capacity and operating parameters.

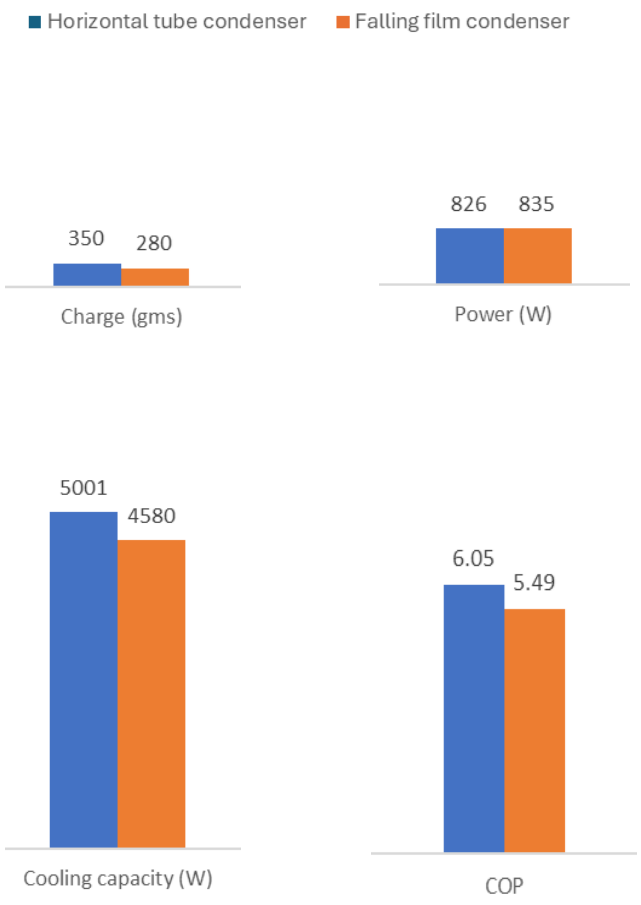


Figure 11. Experimental result comparison.

6.3 Experiment Results

A comparison was conducted between a conventional horizontal tube condenser (5 mm outside diameter) and a falling film condenser (7 mm outside diameter). Figure 11 captures the parametric comparison of the two cases. While the falling film design achieved a 20% reduction in charge inventory, it resulted in an 8.4% reduction in cooling capacity and a 1% increase in power consumption. Simulations suggest that further charge reduction is possible by utilizing smaller tube diameters in falling film designs

7.0 Conclusions

The study concludes that HC-290 (propane) serves as a viable, natural, low-GWP alternative to high-GWP HFC refrigerants, which are currently being phased down under the Kigali amendment. To comply with safety standards such as EN 378 that restrict propane charge, the research successfully implemented a falling film condenser design to minimize refrigerant inventory while nearly maintaining the system's cooling capacity and COP. Simulated results using a MATLAB model indicated a 24% reduction in refrigerant charge compared to the base case, with only marginal variations in performance.

Experimental studies involving a larger tube diameter showed a 20% reduction in charge when the falling film condenser was compared to a traditional horizontal tube condenser.

Ultimately, it proves it is possible to significantly reduce refrigerant charge without negatively affecting the performance of the air conditioner. Future efforts will involve conducting experiments with smaller tube sizes to further optimize the design

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Nomenclature

p	pressure (kPa)	Nu	Nusselt number
T	temperature (K or ^{0}C)	h	Heat transfer coefficient ($Wxm^{-2}xK^{-1}$)
U	Overall Heat transfer coefficient ($Wxm^{-2}xK^{-1}$)	g	Gravitational acceleration ($9.81\ mxs^{-2}$)
r	Latent heat ($Jxkg^{-1}\ xK^{-1}$)		
λ	Thermal conductivity ($Wxm^{-1}\ xK^{-1}$)	μ	Dynamic Viscosity (Pa-s)

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